GDC-63-197

DEVELOPMENT OF A HIGH-TEMPERATURE NUCLEAR-RADIATION-RESISTANT PNEUMATIC POWER SYSTEM FOR FLIGHT VEHICLES

AF 33(616)-7582

QUARTERLY REPORT

24 September 1963

NOTICE

The information contained herein is advanced information and has not been approved by ASD. The USAF assumes no responsibility for the information or conclusions presented.

FOREWORD

This report, prepared by General Dynamics/Convair, covers research and development work accomplished under Air Force Contract AF 33(616)-7582 between 25 June 1963 and 24 September 1963.

The contract was initiated under Project Task No. 61085 by the Aeronautical Systems Division, Wright-Patterson Air Force Base, Ohio. The work is administered under the direction of Mr. B. P. Brooks, APIP-30, Power Conversion and Conditioning Branch, Flight Vehicle Power Division of the Aero-Propulsion Laboratory. It is conducted by Convair under the direction of Mr. R. W. Casebolt.

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INTRODUCTION

The development of a high-temperature, nuclear-radiation-resistant pneumatic power system for flight vehicles was initiated by the Aeronautical Systems Division of the U.S. Air Force to advance the state of the art of pneumatic systems.

Future flight vehicles will be subjected to extreme thermal environments because of heat generated by supersonic speeds and/or the propulsion systems. This results in new requirements for components and auxiliary systems. Therefore, it is a goal of this program to design, develop, and test a typical pneumatic system capable of operating under wide temperature excursions while in the presence of nuclear radiation.

The work accomplished during the first portion of the program indicated that subsequent effort should be directed toward the development and testing of a closed, high-pressure pneumatic system for powering aerodynamic controls. A ram air turbine-driven compressor was selected as the most feasible pneumatic power source.

In addition to the compressor, other components currently under development include a rotary actuator and servo control valve, pressure regulator, relief valve, accumulator, filter, and check valve as well as miscellaneous hardware. Subsequent to evaluation, these components will be tested as a pneumatic system under a simulated high-temperature environment.

SUMMARY

The preliminary study for Phase II of the turbocompressor program was completed; a satisfactory configuration was established for a four-stage centrifugal compressor utilizing a single shaft. Three high-temperature (1,500° F) experimental test runs were conducted on the pilot portion of the Model X pressure regulator while experimental room temperature testing of the Model F relief valves also was completed with satisfactory results.

In regard to the Rotary Actuator and Servo package (RASP), inadequate penetration of the brazing alloy used by the subcontractor as well as fabrication problems have resulted in some delay to the test program. A repair scheme was initiated to replace a defective volume chamber on this unit.

1 WORK ACCOMPLISHED

1.1 TURBOCOMPRESSOR

The preliminary study was completed for Phase II, Design and Experimental Testing of the Turbocompressor. Feasibility was established for supporting all four compressor stages on a single shaft to provide a 4:1 compression ratio (500 psia to 2,000 psia).

The basic approach was to determine what changes could be made to the two-section machine to reduce impeller mass. A computer analysis was then undertaken to determine the dynamic behavior of the rotating group supported on a step-type air bearing. Three different configurations were investigated as follows (see Figure 1):

a. Configuration A — The four centrifugal impellers of the Phase I design, originally on two separate shafts, were rearranged as shown. A critical speed analysis of this configuration indicated potential failure at 69,000 rpm. This design, therefore, is not considered feasible for use in the Phase II machine.

b. Configuration B — Critical speed and stability analyses have shown this unit to be satisfactory for the intended application. It is submitted as the design that best satisfies the requirements. Aerodynamic considerations required some modification of the impeller design in favor of integral shrouding.

c. Configuration C — Critical speed and stability analyses showed this unit also was satisfactory for a four-stage, single-shaft compressor

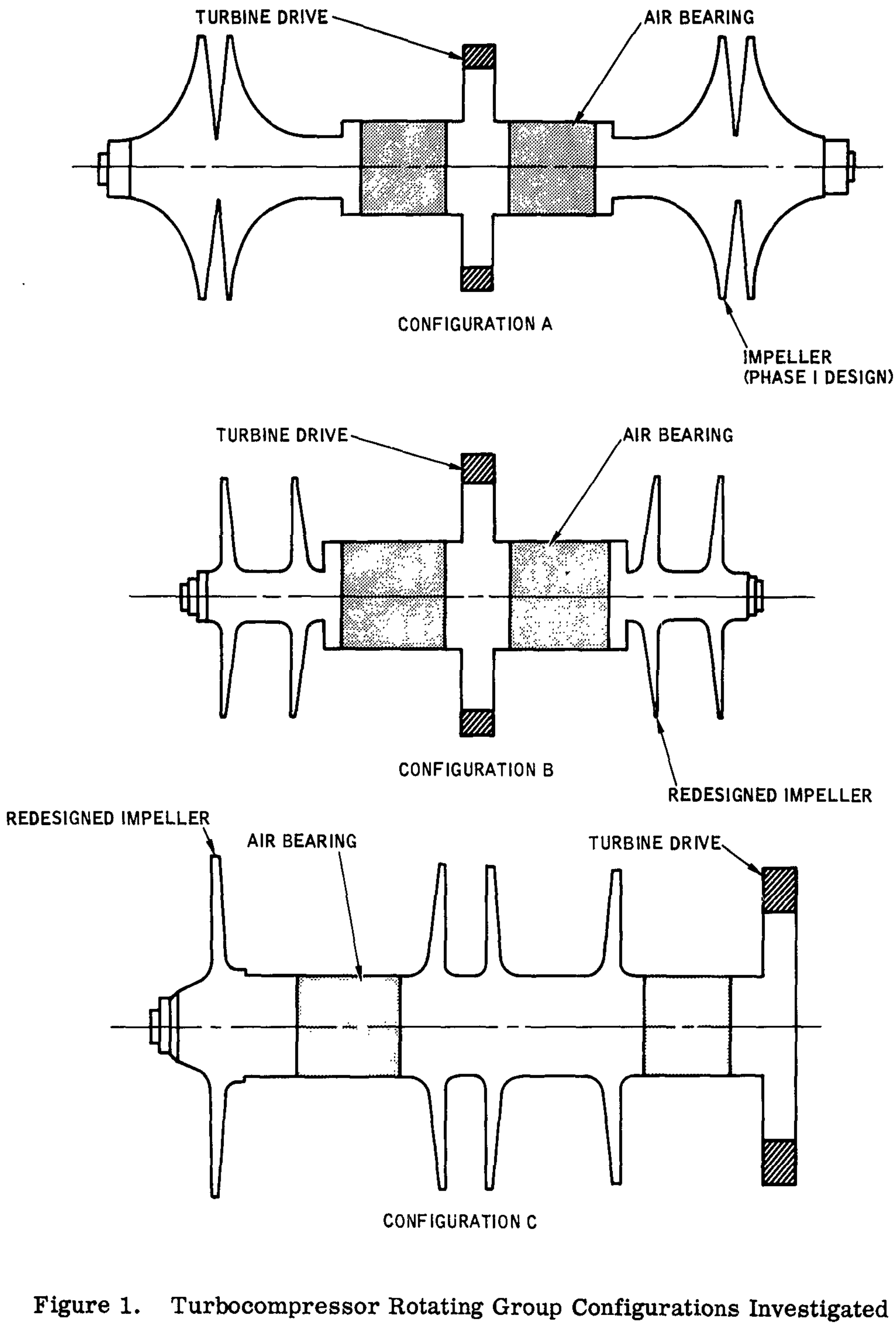

Figure 1. Turbocompressor Rotating Group Configurations Investigated

unit. However, further consideration revealed that this design would present fabrication and assembly difficulties. Therefore, Configuration C was abandoned.

To improve the stability of Configuration A, both the mass and distance of each impeller from the bearing location was reduced, resulting in Configuration B. Two approaches were used in reducing the mass of the impeller: (1) increased rotational speed and (2) elimination of axial inducer section of impeller.

Figure 2 shows the result of a stability analysis on the rotor-bearing system of Configuration A. Superimposed on this graph are the performance characteristics of the 1.25-in.-diameter air bearing previously proposed for Phase II (see Figure 9, Ref. 1). At 100,000 rpm, a zero eccentricity gas film stiffness of 2,000,000 lb./in, would be required to support the rotor (neglecting the possibility of failure at 69,000 rpm). The rotor that was experimentally evaluated during Phase I had a diameter of 1,875 in. and exhibited a stiffness of 800,000 lb./in. For a 1.25-in.-diameter shaft, the comparable stiffness would be 530,000 lb./in. Thus, it is concluded that the bearing design required for Configuration A is far beyond that which has been accomplished to date.

Figures 3, 4, and 5 show the results of the rotor bearing stability analyses of Configuration B without shrouding while Figure 6 is the result with shrouding. Superimposed on each of these graphs are the performance characteristics of the 1.25-in.-dia. air bearing proposed previously for Phase II. In all cases, the minimum stiffness required to ensure freedom from half-frequency whirl instability at the design speed (115,000 rpm) is less than the equivalent 530,000 lb./in. which was experimentally demonstrated during Phase I. Although each of these systems (Figures 3 through 6) represents a feasible selection for the rotor bearing, aerodynamic considerations require that the impellers be fully shrouded for expected performance.

A rotor-bearing stability analysis of Configuration C is shown in Figure 7. In this case, the design operating speed is 90,000 rpm. The minimum stiffness

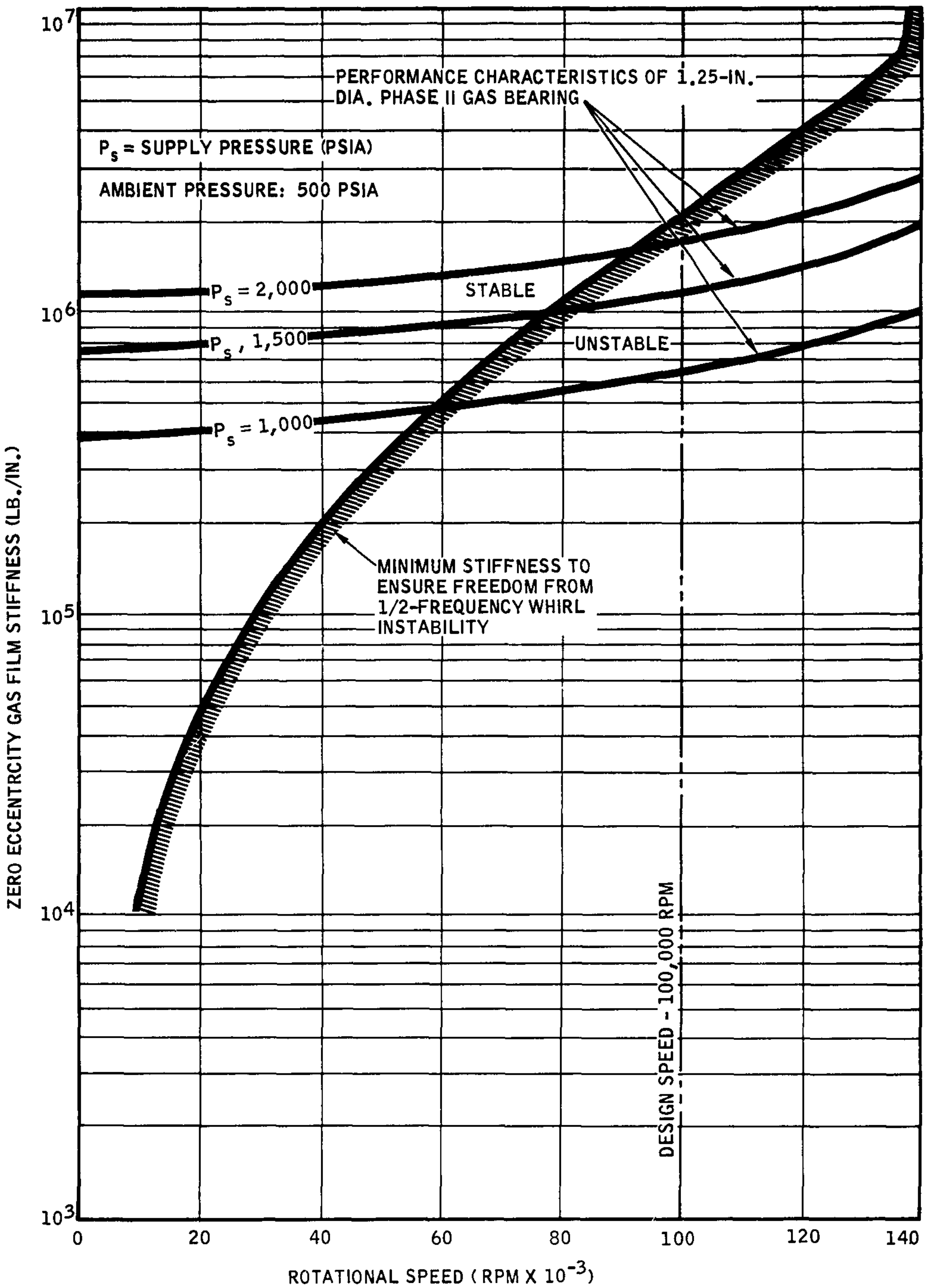

Figure 2. Stability Characteristics of Configuration A Design

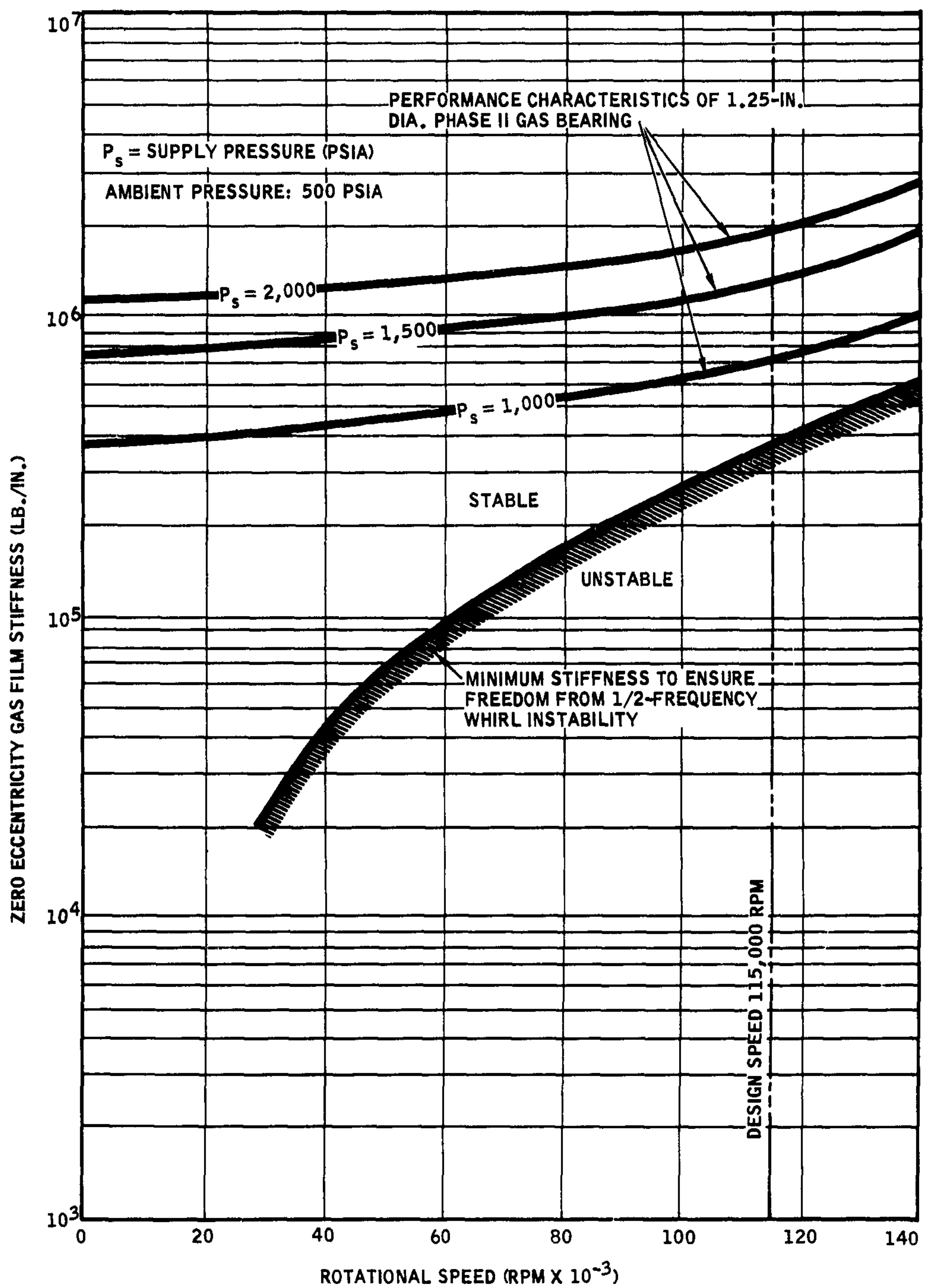

Figure 3. Stability Characteristics of Configuration B Design (0.35-In.-Dia. Impeller Shaft, No Shrouds)

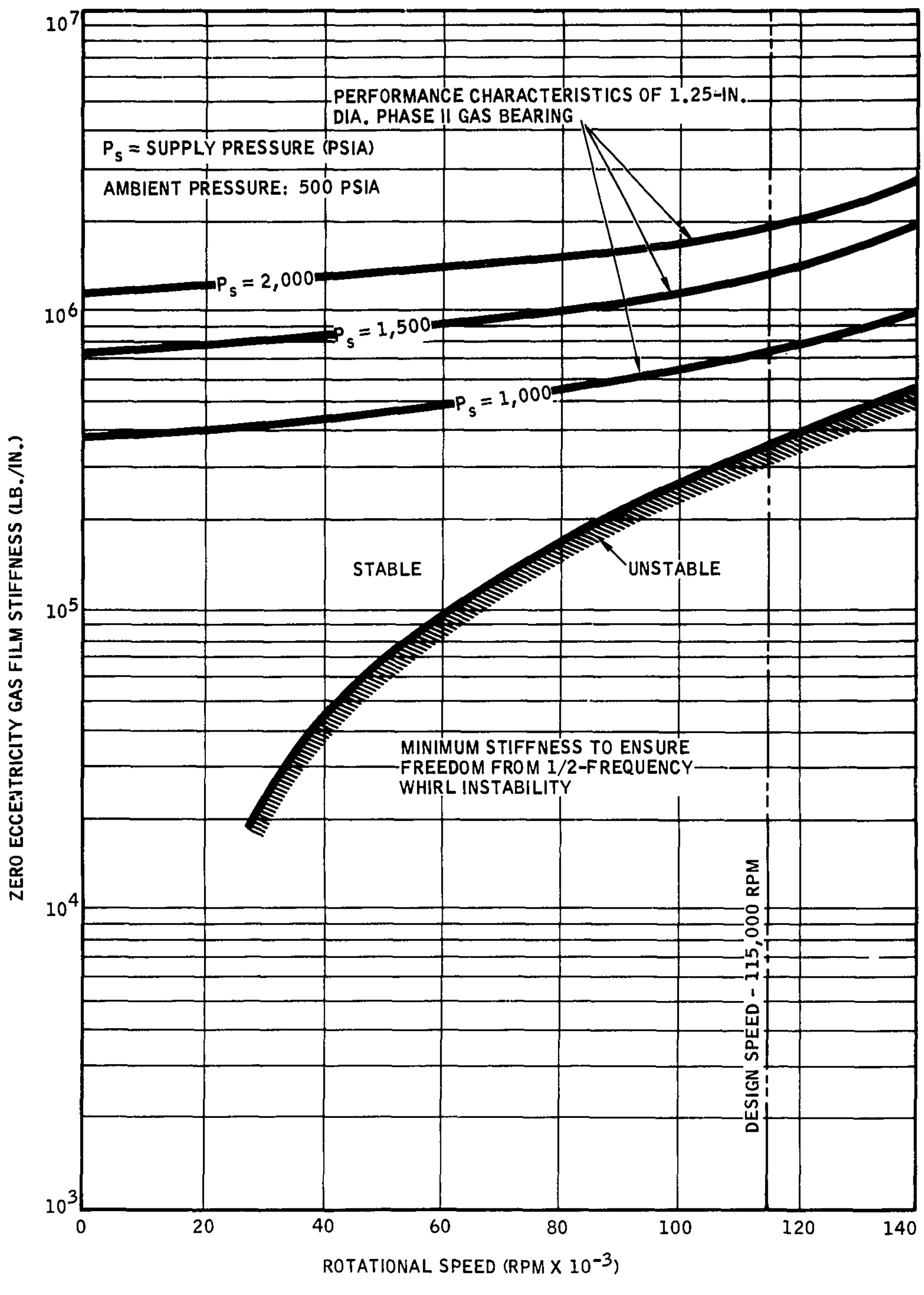

Figure 4. Stability Characteristics of Configuration B Design (0.40-In.-Dia. Impeller Shaft, No Shrouds)

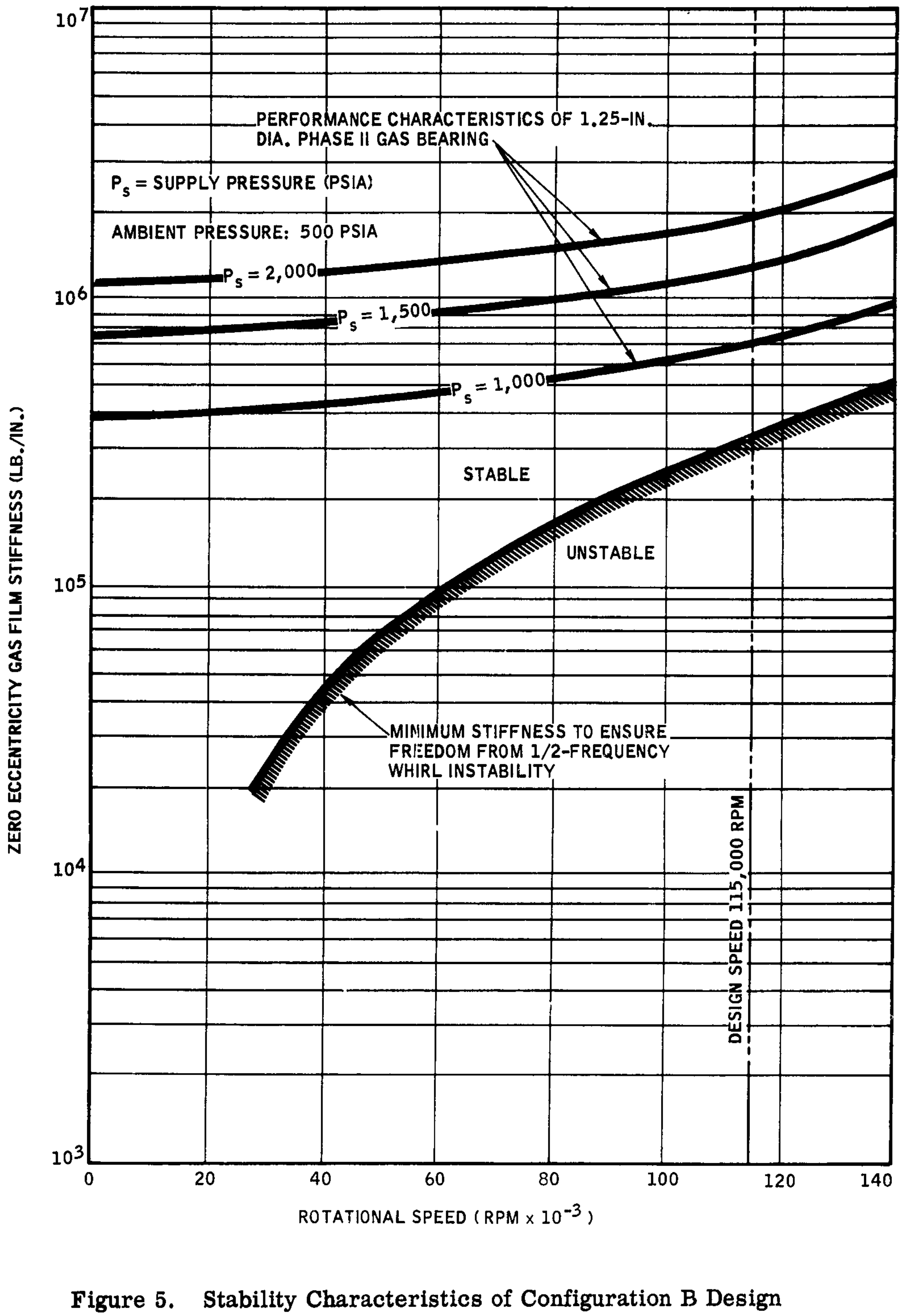

Figure 5. Stability Characteristics of Configuration B Design (0.50-In.-Dia. Impeller Shaft, No Shrouds)

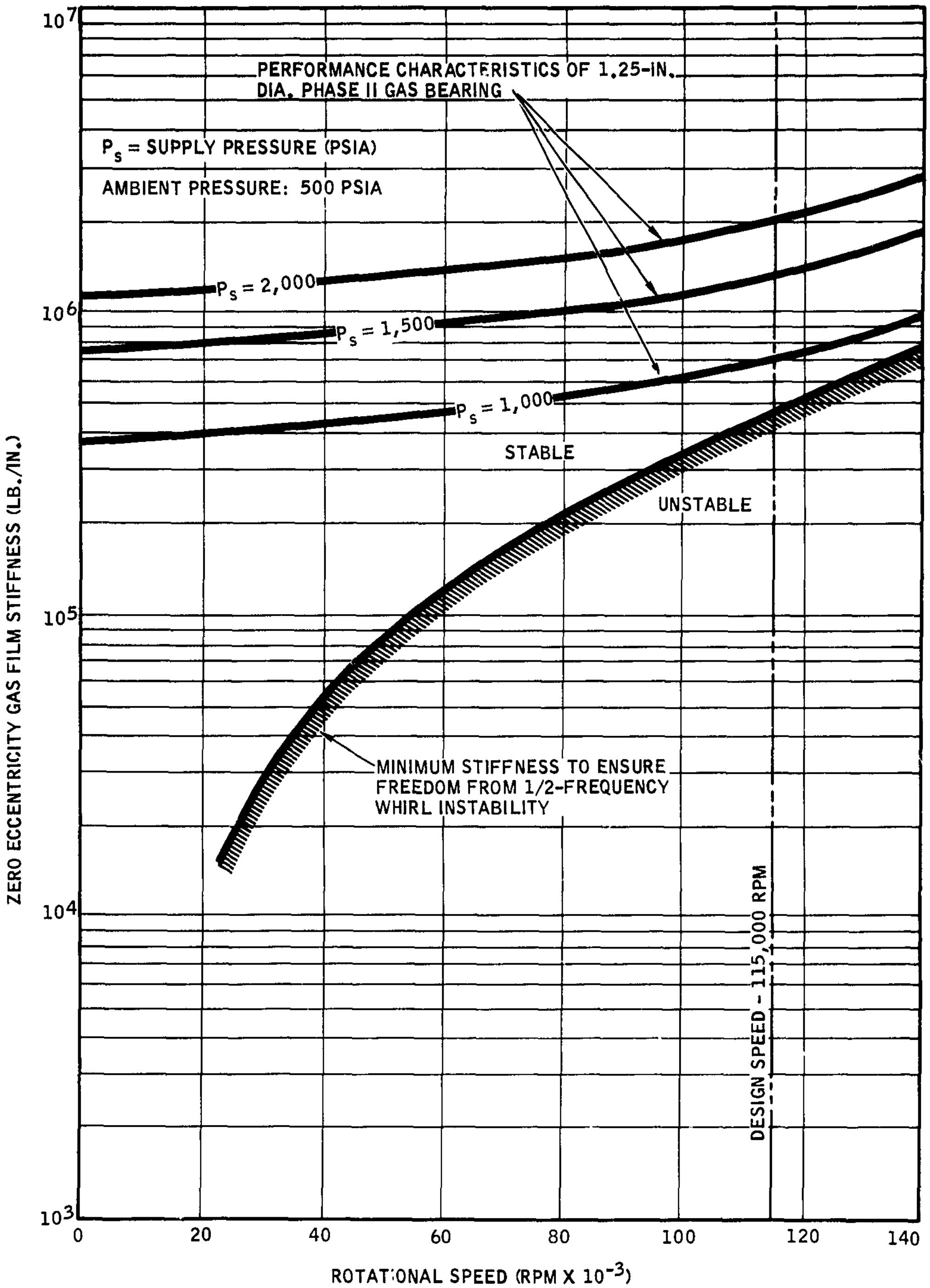

Figure 6. Stability Characteristics of Configuration B Design (0.35-In.-Dia. Impeller Shaft, With Shrouds)

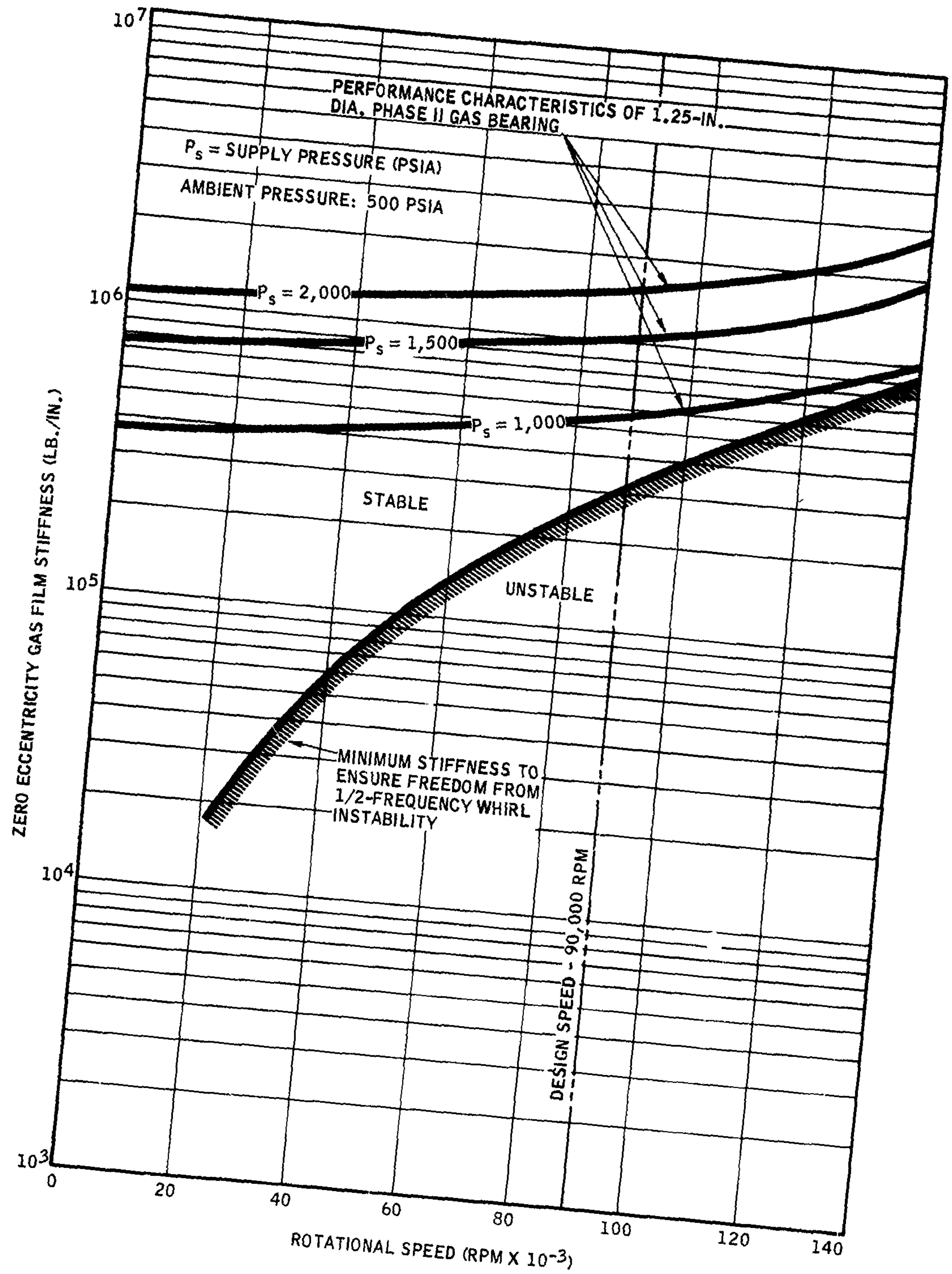

Figure 7. Stability Characteristics of Configuration C Design (No Shrouds)

required to ensure freedom from half-frequency whirl instability is 300,000 lb./in. which again is less than the equivalent 530,000 lb./in. already demonstrated during Phase I. The physical arrangement of Configuration C is not well suited to the placing of shrouds on all the impeller stages. In addition, since fabrication and assembly would be more difficult than Configuration B, further work on the Configuration C design was stopped.

Table 1 is a comparison of the compressor characteristics for the previously proposed two-section machine and the single-shaft machine. Increasing the rotational speed of the unit from 100,000 to 115,000 rpm increases the attainable efficiency of the compressor. The second modification (elimination of the inducer section) will tend to decrease the efficiency of the impeller. The net effect, however, results in a slightly increased efficiency as shown in Table 1. It also should be noted that both the impeller diameter and tip speed will be reduced as a result of the increased rotational speed.

The gas-lubricated journal bearing is identical to that which was finally designed in Phase I (i.e., 1.25-in. dia., externally pressurized, step journal bearing). This preliminary study established that the minimum stiffness required to maintain stability for Configuration B is less than that demonstrated during Phase I. On this basis, feasibility was established for supporting all four compressor stages on a single shaft. Although no changes to the gas-lubricated journal bearing are required, some modification of the impellers will be necessary.

A detailed schedule and work statement for both Phase II alternates is being prepared. This includes both the original two compressor stages/shaft concept and the revised four compressor stages/shaft machine discussed in this report.

1.2 ROTARY ACTUATOR AND SERVO PACKAGE

One RASP was received by Convair and room temperature inspection testing was initiated. A failure of the pressure manifold and volume chamber, due to

Table 1. Comparison of Compressor Characteristics

	TWO-SECTION MACHINE SPEED: 100,000 RPM FLOW RATE: 20 LB./MIN. PRESSURE RATIO: 4:1				SINGLE-SHAFT MACHINE SPEED: 115,000 RPM FLOW RATE: 20 LB./MIN. PRESSURE RATIO: 4:1			
STAGE	1	2	3	4	1	2	3	4
Inlet Temperature (° F)	680	695	737	753	680	695	737	753
Inlet Pressure (psia)	484	764	1,122	1,527	484	764	1,122	1,527
Pressure Ratio	1.60	1.48	1.37	1.31	1.60	1.48	1.37	1.31
Exit Pressure (psia)	774	1,132	1,537	2,000	774	1,132	1,537	2,000
Stage Efficiency	0.48	0.42	0.40	0.37	0.49	0.45	0.43	0.40
Load Factor	15.1	16.3	16.7	17.2	12.9	14.0	14.45	15.0
Impeller Diameter (in.)	3.56	3.48	3.24	3.13	2.91	2.84	2.625	2.54
Tip Speed, (fps)	1,555	1,519	1,413	1,365	1,460	1,425	1,320	1,275

inadequate penetration of the brazing alloy used by the subcontractor, has resulted in some delay to the test program. Although repair of the pressure manifold was completed, two attempts to weld the René 41 volume chamber were unsuccessful. A third attempt is now underway using a revised configuration.

Figure 8 is a photograph of the RASP, in the as-received condition, showing the location of the pressure manifold and transient pressure volume chamber. The RASP was installed in the actuator loading mechanism followed by calibration of instrumentation (see Figures 9 and 10 for two views of the actuator loading mechanism). Proof-pressure testing was then initiated. While a servo signal pressure of 1,000 psig (null pressure) was maintained, both the inlet and return pressures were slowly increased. A failure of the pressure manifold occurred between 800 and 900 psig inlet pressure. Figure 11 is a general view of the installed RASP subsequent to this failure while Figure 12 is a closeup of the pressure manifold. Examination of the cylinder and end cap revealed that little penetration had occurred during the brazing process.

The assembly was repaired by TIG welding, after first chamfering the end cap to increase the weld contact area. Once this repair was completed, a pressure of 50 psig was applied to both the inlet and return ports simultaneously. Although the pressure manifold was then satisfactory, external leakage was discovered at the transient pressure volume (servo signal side of the first stage valve). Stratos-Western Branch, when notified of the preceding developments, made the following recommendations:

a. Remove the defective volume chamber by sawing at the connecting tube junction.

b. Manufacture a new chamber from a 2-in. OD René 41 tube. The total volume (tubing plus chamber) should equal the original volume.

c. Reinstall the new volume chamber in a location which would provide access for rebrazing.

Figure 8. Rotary Actuator and Servo Package (As Received)

Figure 9. Actuator Loading Mechanism Inertia Load

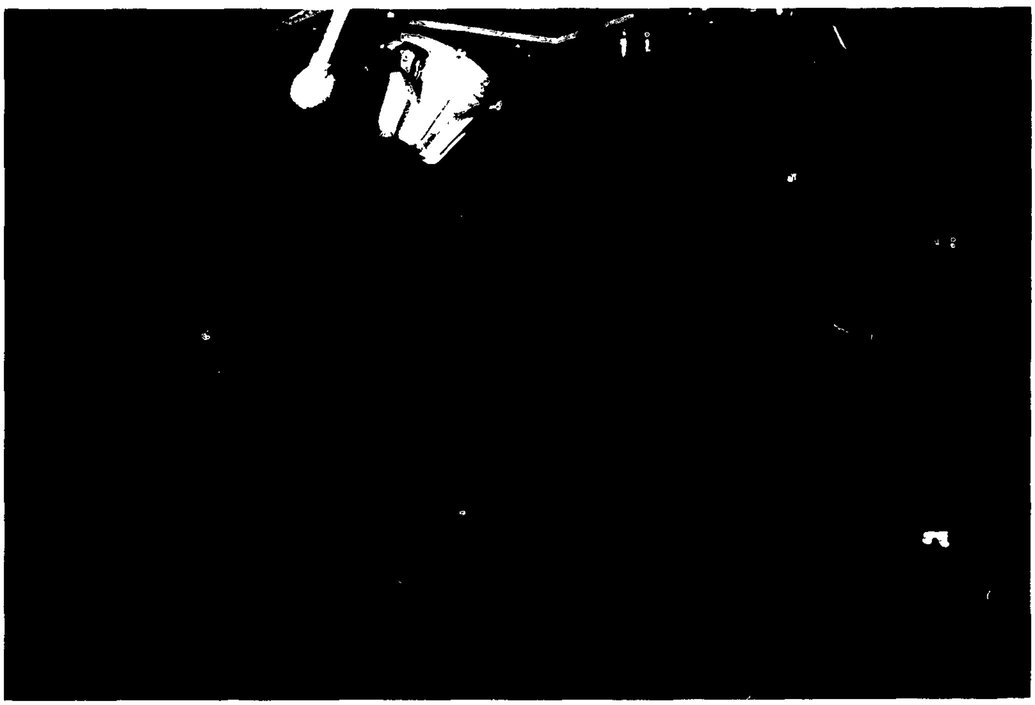

Figure 10. Actuator Loading Mechanism Torsion Bar

Figure 11. RASP Pressure Manifold Failure, General View

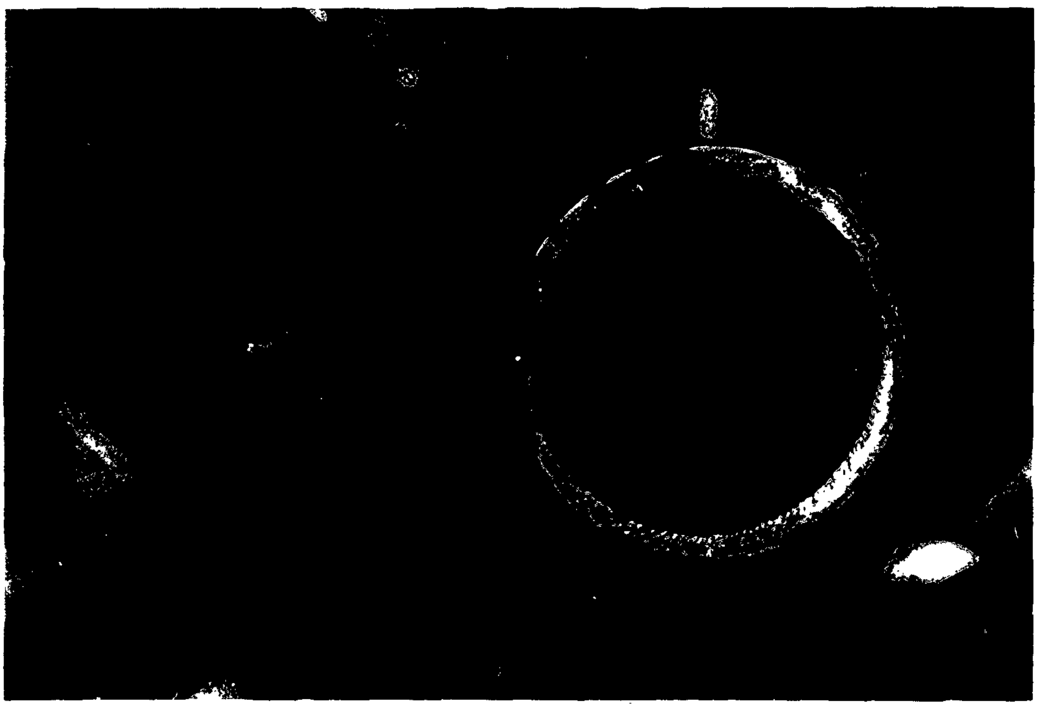

Figure 12. RASP Pressure Manifold Failure, Detail View

The defective volume chamber was removed and a new cylinder manufactured using a 2-in. OD René 41 tube supplied by the subcontractor and available scrap material for the end plates. After welding, the assembly was charged into a hot furnace (to minimize time in the aging temperature range) and stress relieved at 1,600° F/2 hours (minimum). This was followed by pressure testing using a sleeve stub for attachment of a suitable fitting. Failure occurred at 1,200 psig when one end closure sheared adjacent to the weld. Both the end closure sheet stock and second tube were checked for hardness. Hardness was R_c 30 thus indicating both materials were in heat-treated condition before welding. Since welding is recommended in the annealed condition only, a second welding attempt was made.

Before welding, the René 41 material was annealed by heating to 1,975° ±25° F followed by a water quench so that 1,200° F was obtained in less than 4 sec. Hardness checked in this condition was $R_c < 27$ (which is satisfactory). The joint was cleaned and welded using inert-gas-shielded arc with a direct-current power supply. Gas was used for both the arc and backup to prevent oxidation. The assembly was then charged into a hot furnace and given a two-step heat treatment as follows:

a. 1,975° ±25° F/1/2 hour, air cooling.

b. 1,400° ±15° F/16 hours, air cooling.

The assembly then was pressure-tested using water. Failure occurred at 2,000 psig when a pin hole leak developed in the weld zone.

After evaluation of these welding attempts and discussion with Manufacturing Research personnel at Convair, it was concluded that extreme cleanliness of both the joint and the welding rod was required to obtain satisfactory welds. In addition, a stress analysis indicated that deflection of the end plates would overstress the material due to a combination of meridional, circumferential, and radial stresses. The volume chamber was then redesigned as a short cylinder with two hemispherical end caps. This revised configuration will require only

one weld and will eliminate stress concentrations experienced previously.

The initial steps were completed. After welding, the assembly will be heat treated and pressure-tested prior to rebrazing to the existing RASP tubing.

1.3 PRESSURE REGULATOR

The pilot portion of the Model X pressure regulator was assembled and experimental testing was resumed. In addition to room-temperature calibration tests, three high-temperature (1,500° F) runs were completed. The results are summarized in Figure 13.

Calibration of the experimental pilot regulator was performed before and after individual high-temperature runs. In each case, the regulated pressure was recorded for inlet pressures of 1,600 to 2,000 psig and flow rates from 0.019 lb./min. to 0.19 lb./min. The calibration tests are presented in Figure 14 while the three high-temperature runs are shown in Figure 15.

The initial regulated pressure was established as 160 ±4 psig. Run No. 1 was then made with inlet pressure at 1,600 psig and flow at 0.019 lb./min. As shown in Figure 15 (A), the regulated pressure was reduced to 90 psig at 1,500 ° F. This change in the regulated pressure is greater than that anticipated due to change in shear modulus of the spring material. During room temperature calibration, the regulated pressure was recorded as 128 ± 4 psig as shown in Figure 14 (B) which indicates that a permanent set of 32 psi in the regulated pressure had occurred.

The pilot regulator was disassembled and examined for possible damage or distortion. None was noted. The unit then was reassembled and recalibrated to give a regulated pressure of 156.5 ±3.5 psig at room temperature as shown in Figure 14 (B). A second high-temperature run gave results similar to Run No. 1; the regulated pressure was 98 ±6 psig at 1,500° F. Figure 15 (B) shows test results. When the unit was checked at room temperature, the regulated pressure held constant at 143 ±3 psig as shown in Figure 14 (D) thus indicating

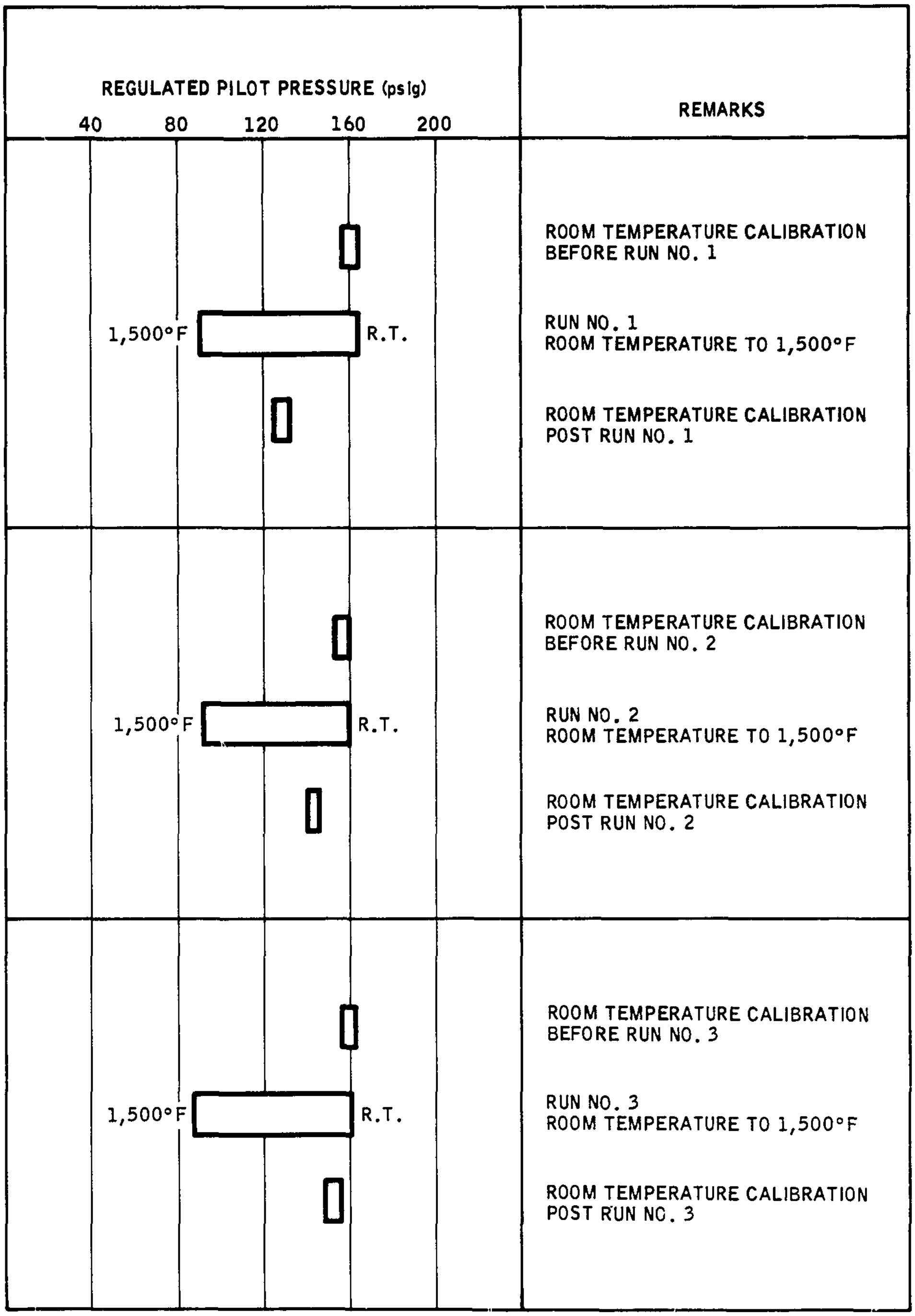

Figure 13. Model X Pilot Pressure Regulator Summary of Test Results

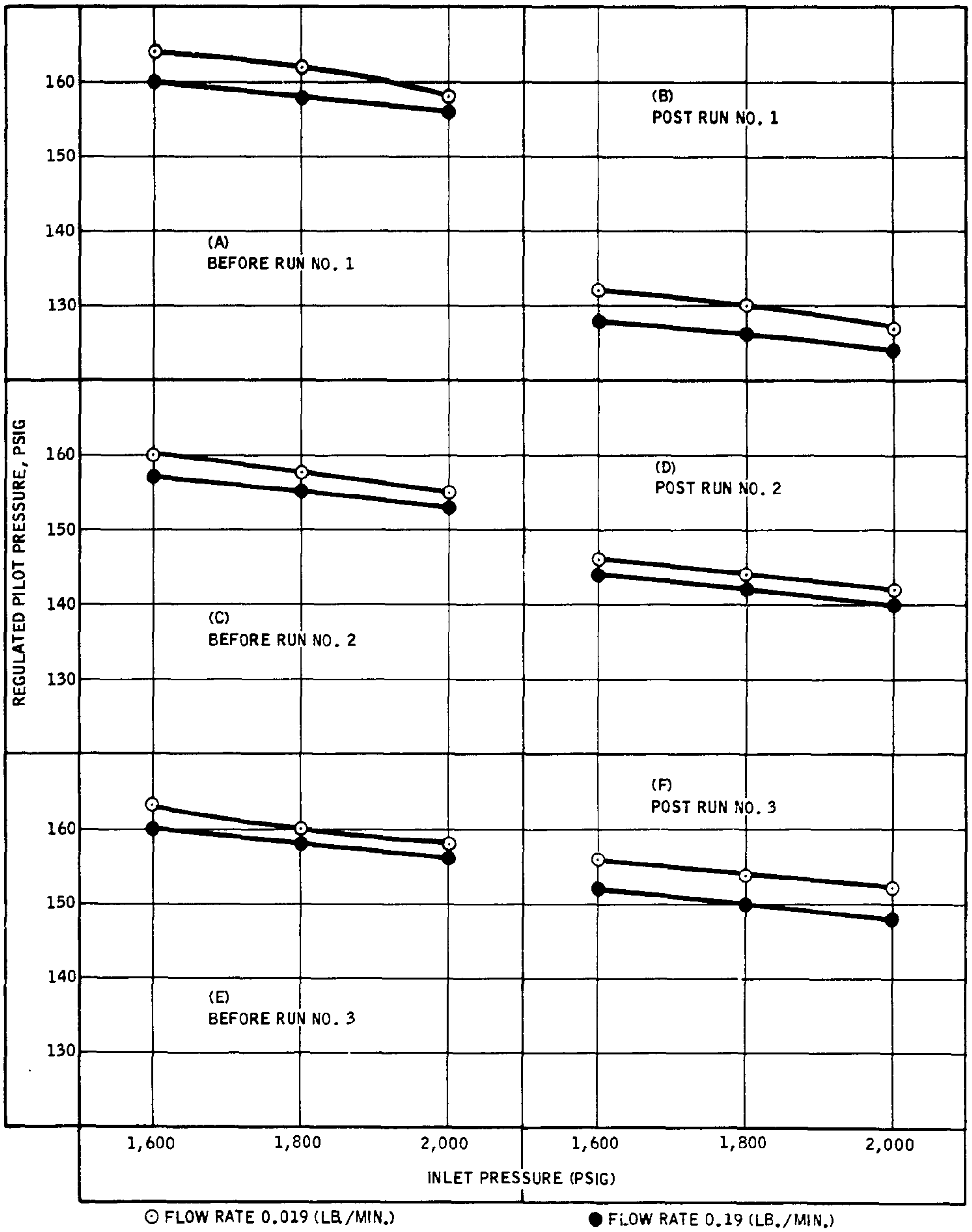

Figure 14. Model X Pilot Pressure Regulator Room Temperature Calibration Tests

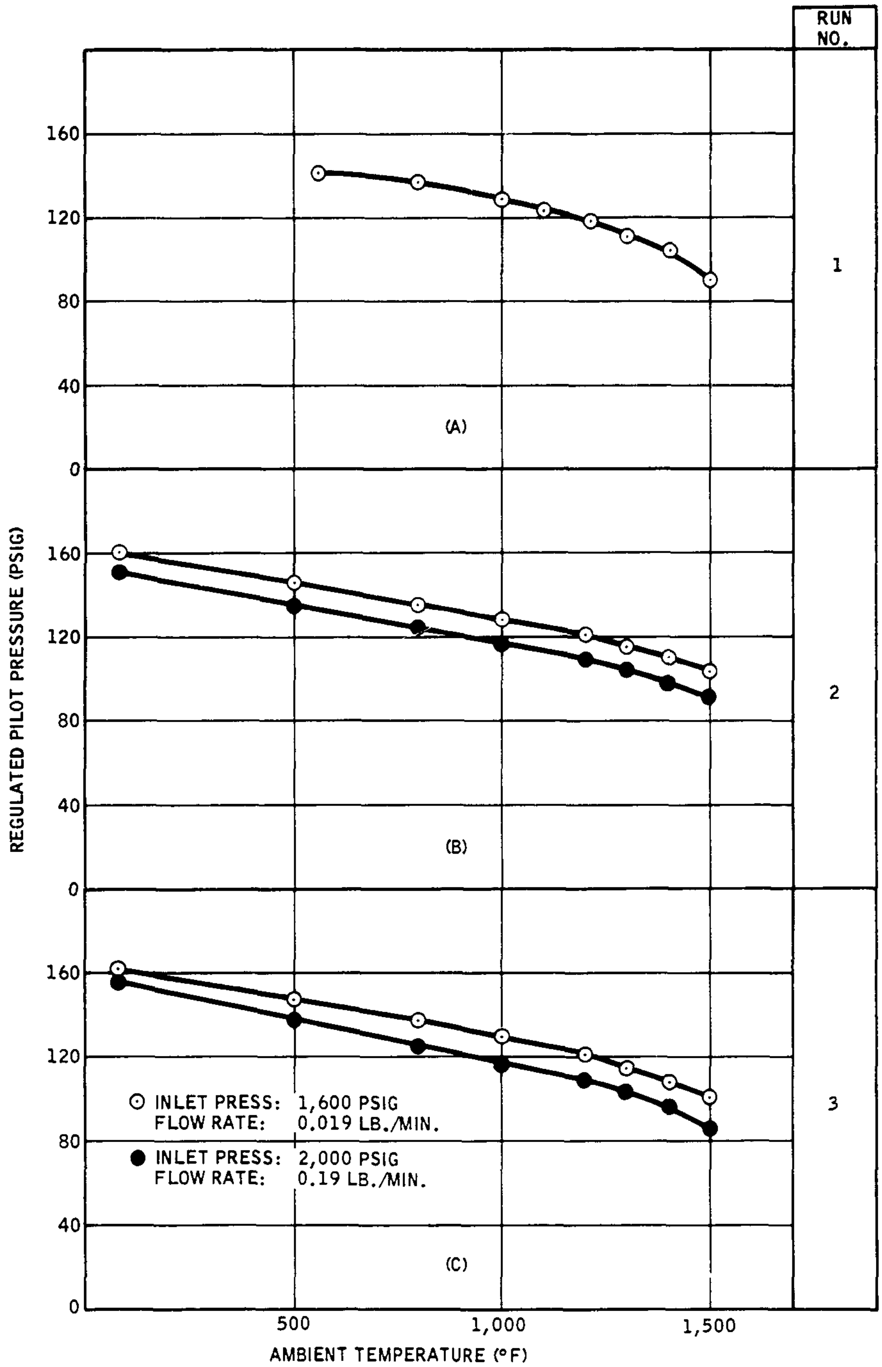

Figure 15. Model X Pilot Pressure Regulator Test Results (Room Temperature to 1,500° F)

that the permanent set had been reduced to 13 psi.

The unit was disassembled and inspected; no damage or deformation was apparent. Examination of the René 41 spring revealed that a permanent set had occurred. Spring height under a test load was given as follows:

Original	1.250 in.
Post Run No. 1	1.220 in.
Post Run No. 2	1.208 in.

Three new springs were fabricated using a revised heat-setting procedure. A constant load (instead of a constant deflection) was maintained as follows: each spring was subjected to 1,600° F for eight hours while supporting a dead weight of 37.5 lb. The deflection of each spring was checked before and after the heat-setting operation.

	LOADED HEIGHT	
SPRING	INITIAL (IN.)	FINAL (IN.)
1	1.250	1.138
2	1.250	1.138
3	1.250	1.103

Spring Number 1 was installed in the unit and a calibration test was completed at room temperature. The unit regulated at 159.5 ±3.5 psig over the inlet pressure range of 1,600 to 2,000 psig and with flows between 0.019 and 0.19 lb./min. as shown in Figure 14 (E). Run Number 3 was then made with results as shown in Figure 15 (C). The unit was recalibrated at room temperature to show the effects of the high-temperature run. Results are shown in Figure 14 (F). The regulated pressure was 152 ±4 psig and the permanent set was reduced to 7.5 psi. Thus it can be seen that the revised heat-setting operation resulted in improved stability of the unit.

When the test unit was removed from the tubing to pipe in controlled pressure to the regulating bellows, two closure screw heads fell off. They had

ruptured part way down the threads, but there was evidence of having been subjected to heating <u>after</u> the rupture. This suggested that they had ruptured during a portion of the high-temperature run. The remaining screws ruptured at the first attempts to remove them.

As soon as new screws are received, the unit will be reassembled and a calibration rerun will be made at room temperature, followed by a run at high temperature with a controlled pressure inside the bellows to maintain constant regulated pressure. A high-temperature run also will be made with preset pressure inside the bellows to determine how closely the unit regulates under this condition.

1.4 RELIEF VALVE

Two Model F relief valves were received by Convair. Figure 16 is a photograph of the unit in the as-received condition. Experimental testing at room temperature was completed and, with the exception of rated flow pressure, the results were within specification limits. In addition, a high-temperature test setup was completed during the period covered by this report.

a. Proof Pressure — The inlet pressure was increased to 2,500 psig while maintaining 1,000 psig at the return port. There was no evidence of failure, excessive distortion, or permanent set.

b. External Leakage — With 1,500 psig at the inlet port and 500 psig at the exhaust port, the external leakage was recorded as follows:

S/N	Leakage
1	0
2	1.06×10^{-5} lb./min.

c. Internal Leakage — The internal leakage was determined at two differential pressures (500 and 1,150 psid) as follows:

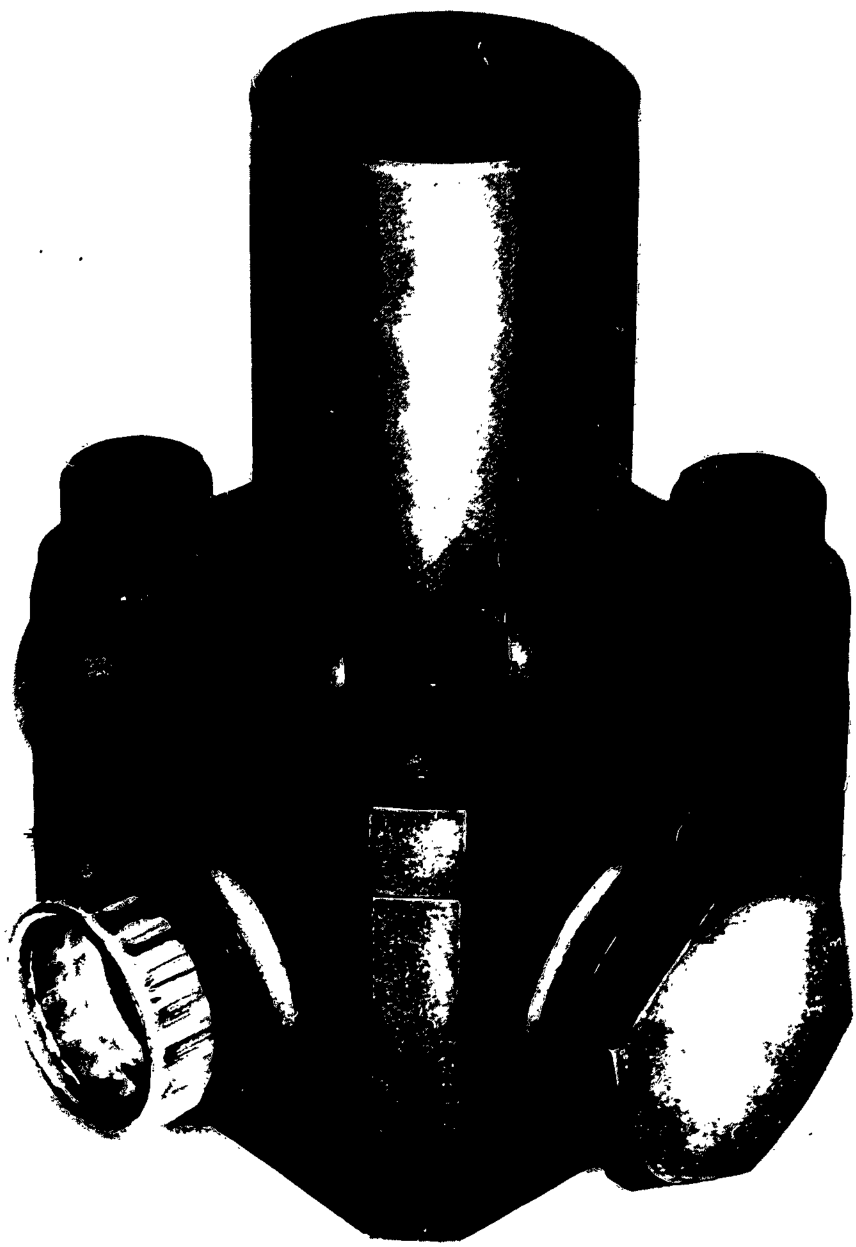

Figure 16. Model F Relief Valve (As Received)

UNIT	PRESSURE (PSID)	LEAKAGE (LB./MIN.)
1	500	2.6×10^{-6}
	1,150	5.8×10^{-5}
2	500	6.6×10^{-6}
	1,150	2.2×10^{-4}

NOTE: Specification requirement is 2×10^{-2} lb./min. (maximum)

d. Operation — While maintaining a back pressure of 500 ±10 psig, the inlet pressure was increased slowly until a sudden increase in flow rate was noted. The inlet pressure was increased further until a flow rate of 20 lb./min. was obtained and rated flow pressure was recorded. The inlet pressure was then decreased slowly, while observing the flow rate; reseat pressure was recorded. The results are shown in Table 2. Although full flow pressure exceeds specification requirement, this will not be detrimental during system testing. In addition it should be noted that the flow (at any given pressure) will tend to increase at the normal operating temperature of 1,500° F due to reduced spring loading.

Table 2. Model F Relief Valve Test Results

UNIT	INLET PRESSURE (PSIG)	RETURN PRESSURE (PSIG)	CRACKING PRESSURE (PSID)	FLOW RATE* (LB./MIN.)	RESEAT PRESSURE (PSID)
1	2,300	500	1,800		
	2,500	0		17.5	
	2,275	500			1,775
2	2,400	500	1,900		
	2,500	0		14.7	
	2,300	500			1,800
CONVAIR SPECIFICATION:			1,300(min.)		1,150 (min.)

*The full flow pressure was not determined since it exceeded proof pressure (2,500 psig).

1.5 FILTER

As reported previously (Ref. 2) excessive external leakage was encountered for Unit No. 1 following the operation test at 1,500° F. This unit was returned to Bendix Filter Division for rework. All attempts to repair the defective weld were unsuccessful. The subcontractor subsequently agreed to manufacture a new filter head subassembly eliminating the weld. Although some delay was experienced in obtaining the René 41 superalloy material, fabrication of the revised configuration is underway with completion scheduled for 27 September 1963.

2 MAJOR PROBLEM AREAS

During the initial stages of this development program it was recognized that a number of problems existed. Resolution of these would, to a great extent, effect satisfactory completion of the program goal — i.e., development of a pneumatic system capable of sustained operation in extreme thermal environments. In general, the following were recognized as major problem areas: springs and bellows, bearings, seals (dynamic and static), material wear properties, and the joining of superalloys. These problems, as well as their solutions, are discussed in this section.

The design of springs and bellows for use at high temperature must be governed by both stress relaxation effects and change of the elastic modulus of the material as a function of temperature. Although springs are still considered a potential problem area, some success was achieved in obtaining satisfactory springs for both the check valve and the relief valve. This was accomplished by using a heat-setting process and operating at a reduced stress level. The maximum anticipated relaxation is 25% after a 70-hour exposure of 1,500° F.

Fabrication difficulties were encountered during construction of the compensator bellows used in the pneumatic pressure regulator. The basic problem, concerned with obtaining satisfactory welds of the ultra-thin (0.003-in.) gage René 41 material, was not resolved. Two bellows of 0.006-in. gage René 41 were fabricated. These have been subjected to repeated thermal cycling with good results.

The development of suitable bearings for extended operation at 1,500° F continues as a possible problem area. Spherical roller bearings were selected

for use in the rotary actuator. Both races and rolling elements were fabricated from René material while the contact surfaces were coated with ceramic-bonded calcium fluoride. Performance of these bearings, evaluated in conjunction with wear testing conducted on a rotary actuator, was judged satisfactory. A gas film lubricated bearing was selected for the compressor. Testing to date has been accomplished at speeds up to 72,000 rpm with satisfactory results.

Seal design is instrumental in determining over-all system efficiency. During the course of this program a satisfactory metal boss seal was evaluated. A piston-ring-type dynamic seal was evaluated during testing conducted on the rotary actuator and the pressure regulator. A total of 888 cycles, over a period of 15 hours at 1,500° F, was completed during wear testing of the rotary actuator. In the case of the pressure regulator, a similar type of dynamic seal was unsatisfactory due to the more stringent leakage requirements for this component.

Some progress was made in resolving the problem associated with material wear properties. The relief valve incorporates a solid chromium-carbide poppet vs. a Haynes alloy No. 25 seat. This combination was successfully evaluated from room temperature to 1,500° F. A ceramic-bonded, calcium-fluoride coating was evaluated during wear testing conducted on the rotary actuator. Although good results were obtained at 1,500° F, an increase in the friction forces was noted at room temperature.

Joining of the superalloys is also considered a major problem area due to difficulties encountered during fabrication of the accumulator and filter elements. A satisfactory weld of the Udiment 500 accumulator was obtained using the electron beam process. In the case of the filter elements, all attempts to vacuum sinter the René 41 filter element were unsuccessful. Good results were subsequently obtained using Hastelloy X wire in combination with a René 41 core.

3 FUTURE WORK

Testing and evaluation of components will continue for subsequent inclusion in the high-temperature pneumatic system. Experimental testing of the RASP will be resumed after completion of the necessary repairs. High-temperature testing of the Model F relief valve and Model X pressure regulator also is scheduled for the next quarterly period.

This experimental testing is required to determine the limitations of pneumatic-type fluid power control systems. Recent advances in materials and techniques are incorporated into these components. Therefore, it is necessary to evaluate each component prior to subjecting the entire pneumatic system to the high-temperature environment.

REFERENCES

1. "Development of a High-Temperature,Nuclear-Radiation-Resistant Pneumatic Power System for Flight Vehicles," General Dynamics/Convair, GDC-63-068, 24 March 1963.

2. "Development of a High-Temperature, Nuclear-Radiation-Resistant Pneumatic Power System for Flight Vehicles," General Dynamics/Convair, GDC-63-140, 24 June 1963.

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